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INTRODUCTION

One of the major noise source in a single stage air compressor is the suction valve. During the compressor operation pulsating air flow is developed in the suction port which interacts with the valve motion resulting in the oscillatory motion of the valve. The magnitude of the oscillating valve velocity determines the capacity of the valve to radiate sound. For reducing the sound power level of the valve, magnitude of the oscillating valve velocity has to be reduced. This is generally achieved by [1, 2],

(i) shaping the valve ports such that separation in air flow occurs over a small fraction of piston motion,

(ii) selecting the size, shape and material of the valve so that its natural frequency is detuned from the range of frequencies of the predominant harmonic components of the pressure pulsations.

Oscillations in the valve motion, and therefore noise generation, can also be reduced by incorporating a valve stop in the suction valve assembly. This is often not done because of the space limitation. In this paper a simplified mathematical model has been developed, from the information available in literature, that predicts the influence of valve stop on the far field sound pressure level and argues for the inclusion of such stops to control noise. Also the model describes the action of a very simple suction muffler which can be made an integral part of the compressor head design.

MATHEMATICAL MODEL

The overall mathematical model consists of

(a) compressor cylinder/valve dynamics model,
(b) suction muffler model,
(c) acoustic model.

Compressor Cylinder/Valve Dynamics Model

The computer program for simulation of positve displacement compressor developed by Soedel and Wolverton [3] was taken as the basic program and modified to predict the valve velocities and volume flow rates through the suction port. The various modifications made are as follows:

A) Using one mode approximation for valve displacement.

Soedel and Wolverton, in their computer program, used first two symmetric modes of the valve in modal expansion to define the displacement of the valve, or

$$W(x,y,t) = \sum_{m=1}^{2} \left(\sum_{i=1}^{n} \phi_{im}(x_i,y_i)\right) q_m(t) \quad (1)$$

where

$W(x,y,t)$ = valve displacement,

$\phi_{im}(x_i,y_i)$ = mode shape of the i^{th} segment of the valve for the m^{th} mode,

$q_m(t)$ = modal participation factor for the m^{th} mode,

n = number of segments the valve is divided into.

In all reasonably well designed valves, the first mode dominates the deflection picture to such an extent that the participation of the higher modes is almost negligible. Only if one is interested in predicting the stresses in valves does the consideration of higher modes become important. As such, equation (1), by using first mode approximation, is simplified to:

$$W(x,y,t) = q(t) \sum_{i=1}^{n} \phi_i(x_i,y_i) \quad (2)$$

(B) Including valve stop in the suction valve assembly.

The second modification made on the Soedel-Wolverton program is inclusion of a valve stop in the suction valve assembly. The shape of the valve stop is either assumed to be the same as the fundamental mode shape of the suction valve or it is assumed that the valve is stiff enough to retain its fundamental mode shape when any point of it contacts the valve stop.

As the piston travels from the top dead center to the bottom dead center, the air pressure in the cylinder gradually decreases. When the pressure in the cylinder P_c, drops below the pressure in the suction muffler, P_s, the suction valve opens; its displacement gradually increases till it touches the valve stop. The modal participation factor of the suction valve, when the valve is between the valve plate and the valve stop, is given by the second order differential equation [4]:

$$\ddot{q}(t) + 2\zeta\,\omega\,\dot{q}(t) + \omega^2\,q(t) = \frac{\Delta P(t)\sum_{i=1}^{n}[\phi(x_i,y_i)\,B\{W(x_i,y_i)\}\Delta A_i]}{A\rho h\iint_s \phi^2(x,y)\,ds} \quad \ldots.(3)$$

where,

ζ = modal damping ratio of valve material

$\Delta P(t)$ = pressure differential across the valve,

$B\{W(x_i,y_i)\}$ = effective force area at the location x_i, y_i (i^{th} segment),

ΔA_i = area of the i^{th} port segment,

A = total port area,

ρ = density of the valve material,

h = valve thickness,

ω = fundamental frequency of the valve,

ds = incremental area of the valve.

Solving the equation (3) for modal participation factor, $q(t)$, the displacement of the valve, $W(x,y,t)$, is calculated by using equation (2).

At the instant the valve touches the valve stop, its acceleration and velocity become equal to zero. The modal participation factor, $q(t)$, at this instant is equal to $q_s(t)$, where,

$$q_s(t) = \frac{W_s(x_i,y_i)}{\phi(x_i,y_i)} \qquad (4)$$

and $W_s(x_i,y_i)$ is the valve stop height at the i^{th} segment.

Substituting equation (4) in equation (3) and setting the acceleration and velocity terms equal to zero, gives the pressure differential, $\Delta P_r(t)$, on the valve which will keep the valve touching the valve stop,

$$\Delta P_r(t) = \frac{\omega^2 q_s(t)\,A\rho h\iint_s \phi^2(x,y)\,ds}{\sum_{i=1}^{n}[\phi(x_i,y_i)\,B\{W_s(x_i,y_i)\}\Delta A_i} \qquad (5)$$

As long as the pressure differential $(P_s - P_c)$ is greater than or equal to $\Delta P_r(t)$, the valve is forced against the valve stop. In the computer program the valve displacement is calculated by the logic statements given in Table 1.

Suction Muffler Model

In many designs, the suction valve is open to the atmosphere except possibly for an air filter. Here we assume that the volume before the suction valve (suction muffler) is separated from the surrounding air by a neck-like restriction. The suction muffler is modeled by treating it as a Helmholtz resonator. The Helmholtz model diagram is shown in Figure 1, where,

dP_s = acoustic pressure

V_{os} = muffler volume

L_s = length of the gas slug which vibrates in the muffler neck

ρ_{os} = mean density of air in the muffler

T_s = mean temperature of air in the muffler

ξ_s = displacement of the gas slug

For detailed analysis of Helmholtz resonator approach for compressor modeling, the reader is referred to reference [5].

Figure 2 shows the various forces acting on the free body diagram of the gas slug, where,

A_s = cross-sectional area of the gas slug

C_s = damping co-efficient

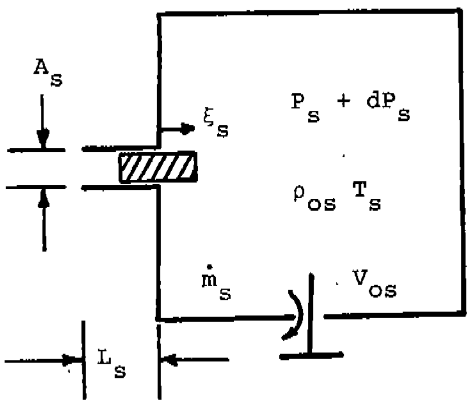

Figure 1. Helmholtz Model of Suction System

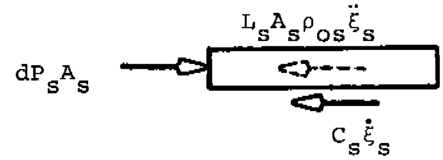

Figure 2. Free Body Diagram of Gas Slug

Force balance on the gas slug gives,

$$L_sA_s\rho_{os}\ddot{\xi}_s + C_s\dot{\xi}_s + dP_sA_s = 0 \tag{6}$$

A positive displacement of the gas slug, ξ_s, will cause a restraining pressure force, given by:

$$dP_sA_s = -\frac{K\,dV_s}{V_{os}} \tag{7}$$

where

dV_s = effective change in volume as seen by the gas slug

K = bulk modulus of air

The value of K can be approximated by [5],

$$K \simeq \rho_{os}C_{os}^2 \tag{8}$$

where

C_{os} = speed of sound in air under the conditions existing in the suction muffler.

The effective change in the volume, dV_s, is given by

$$dV_s = \int_0^t Q_s dt - \xi_sA_s \tag{9}$$

where

t = time elapsed

Q_s = volumetric flow rate into suction port

Substituting equations (7), (8) and (9) in equation (6) gives

$$\ddot{\xi}_s + 2\zeta_s\omega_{ns}\dot{\xi}_s + \omega_{ns}^2\xi_s = R_s\int_0^t \dot{m}_s dt \tag{10}$$

where

$$R_s = \frac{C_{os}^2}{V_{os}\rho_{os}L_s} \tag{11}$$

$$\omega_{ns} = C_{os}\sqrt{\frac{A_s}{V_{os}L_s}} \tag{12}$$

$$\zeta_s = \frac{C_{os}}{2L_sA_s\rho_{os}\omega_{ns}} \tag{13}$$

$$\dot{m}_s = Q_s\rho_{os} \tag{14}$$

$$C_{os} = \sqrt{kgRT_{os}} \tag{15}$$

k = specific heat ratio

g = gravitational constant

R = gas constant

Also solving equations (7), (8) and (9) for acoustic pressure in the suction muffler, gives

$$dP_s = \frac{C_{os}^2}{V_{os}} \left[\rho_{os}\xi_s A_s - \int_0^t \dot{m}_s dt \right] \quad (16)$$

Equations (10) and (16) have to be solved simultaneously with valve dynamics equations to predict the displacement of the gas slug, ξ_s, and the acoustic pressure in the suction muffler, dP_s, at any instant of time.

Acoustic Model

The volume flow rate at the inlet neck of the suction muffler, $(\dot{\xi}_s A_s)$, is a periodic wave in the time domain of a period equal to $2\pi/\omega_c$ seconds, where ω_c is the angular velocity of the compressor shaft in radians per second. By carrying out the harmonic analysis in the computer program, the frequency spectrum of the volume flow rate at the inlet neck of the suction muffler is developed. The frequency spectrum gives the amplitude, Q_m, of the volumetric flow rate at a series of frequencies ω_m, which are the integral multiple of the frequency of the compressor crank shaft, ω_c.

For predicting the acoustic pressure in the far field, the oscillating gas slug at the inlet of the suction muffler can be treated as a monopole source [6] generating spherical acoustic waves. Under this assumption, the magnitude of the acoustic pressure, P_{a_m}, at a distance r in the far field is given by [6],

$$P_{a_m} = \frac{\rho_o \omega_m Q_m}{4\pi r} \quad (17)$$

where

$$\omega_m = m\omega_c \quad (18)$$

m = frequency numbers (1,2,3 ...)

Q_m = frequency content of the volumetric flow rate of the gas slug at ω_m frequency

P_{a_m} = acoustic pressure at frequency ω_m

ρ_o = density of air surrounding the compressor

The sound pressure level, SPL, at various frequencies, ω_m, is given by

$$SPL = 20 \log_{10} (P_{a_m}/P_{ref}) \quad (19)$$

where

P_{ref} = reference effective pressure

= 0.0002 Newtons/m^2

NUMERICAL SOLUTION

The overall mathematical model consists of a set of differential and algebraic equations. The differential equations have been solved by fourth order Runge-Kutta method. This involves solving the algebraic equations five times; four times during the Runge-Kutta procedure and once after the solution of the differential equations at the end of each time step.

In the Soedel-Wolverton computer program, the valve dynamics differential equations are not solved in case both the suction and discharge valves are closed during any particular portion of the compression and expansion cycle. This has been done to achieve minimum computational time since the solution of these differential equations under the above conditions would furnish the trivial result of zero displacement. However, even if both suction and discharge valves are in closed position at any instant of time, the gas slug at the inlet of the suction muffler will still have displacement and velocity. Since in Runge-Kutta routine, the differential equations are solved simultaneously, care has to be taken that the differential equations associated with the gas slug are solved at all times even when both the suction and the discharge valves are closed.

RESULTS AND DISCUSSION

The mathematical model developed above was applied to a single stage air compressor. The input data for the computer program was taken from reference [3].

The first computer program run was made with the suction valve open to the atmosphere and the valve stop excluded from the suction valve assembly. Figure 3 shows the sound pressure level (SPL) spectrum at a distance of 2 meters from the compressor suction valve. The maximum SPL of 85 decibels occurs at 465 rads/sec which is also the bending natural frequency of the suction valve. The other resonance peaks occur in the vicinity of the integral multiples of the fundamental frequency of the suction valve. These peaks in the SPL curve are due to the suction valve flutter which is shown in figure 4 where suction valve displacement is plotted against the crank angle.

The second program run was made by including a valve stop in the suction valve assembly and again keeping the suction valve open to the atmosphere. Figure 5 shows the suction valve displacement plotted against crank angle for this case. It is seen from this figure that the valve flutter present in figure 4 is completely eliminated. This results in appreciable reduction of sound pressure level at frequencies which are integral multiples of the fundamental frequency of the suction valve. Figure 6 shows the SPL spectrum for this case. By comparing figures 5 and 6 it is evident that the inclusion of valve stop greatly reduces the overall sound pressure level.

The third program run was made by including a suction muffler and removing the valve stop from the suction valve assembly. Figure 7 shows the sound pressure level spectrum for this case. The sound pressure level is reduced by about 6 to 10 dB for frequencies above 400 rad/sec. However, the SPL curve still peaks at frequencies which are integral multiple of the suction valve bending frequency. This is because the flutter of the suction valve can not be eliminated by including a suction muffler.

The final program run was made by including both the suction muffler and the suction valve stop. Figure 8 shows the sound pressure level spectrum for this case. As in Figure 6, the peaks in SPL curve which ar associated with the valve flutter are eliminated and the sound pressure level is further reduced by 2 to 5 dB for frequencies above 400 radians/sec.

CONCLUSION

It is concluded from the above study that inclusion of a valve stop and a suction muffler appreciably reduces the sound pressure level. Although it is desirable to include both the suction muffler and the valve stop in the compressor, for economic reasons a choice might have to be made between the two. Though valve stop and the suction muffler individually reduce the sound pressure level significantly, the valve stop has an added advantage in the sense that the peaks in the SPL spectrum, which occur due to the flutter of the valve, are completely eliminated. This results in a large reduction in overall sound pressure level.

LIST OF REFERENCES

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NOMENCLATURE

A	Suction port area (m^2)
A_s	Cross-section area the gas slug at inlet of the suction muffler (m^2)
$B\{W(x_i, y_i)\}$	Effective force area at the i^{th} segment of the valve (m^2)
C_{os}	Speed of sound in the suction muffler (m/sec)
C_s	Damping coefficient of the valve material (N sec/m)
dP_s	Acoustic pressure in the suction muffler (N/m^2)
dv_s	Effective change in volume of the suction muffler (m^3)
g	Gravitational constant [m/sec^2]
h	Valve thickness (m)
k	Specific heat ratio
K	Bulk modulus (N/m^2)
L_s	Length of gas slug at the inlet of the suction muffler (m)
$\dot{m}_s$	Mass flow rate of air through the suction port ((kg/sec = N sec/m)
n	Number of segments valve is divided into
P_a	Acoustic pressure in the medium surrounding the compressor (N/m^2)
P_c	Absolute pressure in the compressor cylinder (N/m^2)
P_{ref}	Reference effective acoustic pressure (N/m^2)
P_s	Absolute pressure in the suction muffler (N/m^2)

$q(t)$	Modal participation factor of the suction valve (m)
Q_m	Volume flow rate in the frequency domain (m^3/sec)
Q_s	Volume flow rate through the suction port (m^3/sec)
r	Distance from the compressor (m)
R	Gas constant (m/°K)
R_s	Effective force on the gas slug per kg of flow (m/sec^2)
S	Surface area of the valve (m^2)
SPL	Sound pressure level (Decibels)
T_s	Temperature in suction muffler (°K)
V_{os}	Volume of suction muffler (m^3)
$W(x,y,t)$	Displacement of the valve (m)
ΔA_i	Area of the i^{th} segment of the suction port (m^2)
ζ	Modal damping ratio of the valve material
ζ_s	Damping ratio of air in the suction muffler.
ξ	Displacement of the gas slug (m)
ρ_o	Density of air surrounding the compressor ($kg/m^3 = N\ sec^2/m^4$)
ρ_{os}	Density of air in the suction muffler ($kg/m^3 = N\ sec^2/m^4$)
$\phi(x,y)$	Fundamental mode shape of the valve
ω	Fundamental frequency of the valve (rad/sec)
ω_c	Rotational frequency of compressor shaft (rad/sec)
ω_{ns}	Helmholtz's resonator frequency of suction muffler (rad/sec)

Displacement of the valve at the previous time step $W(x,y,t)$	Pressure differential across the valve $(P_s - P_c)$	New displacement of the valve $W(x,y,t)$
Zero (Valve touching the valve plate)	Less or equal to zero	Zero
	Greater than zero	Use equations (3) and (2)
Greater than zero and less than valve stop height	-	Use equations (3) and (2)
Equal to valve stop height	Greater or equal to $\Delta P_r(t)$	Zero
	Less than $\Delta P_r(t)$	Use equations (3) and (2)

Table 1. Valve Displacement Logic

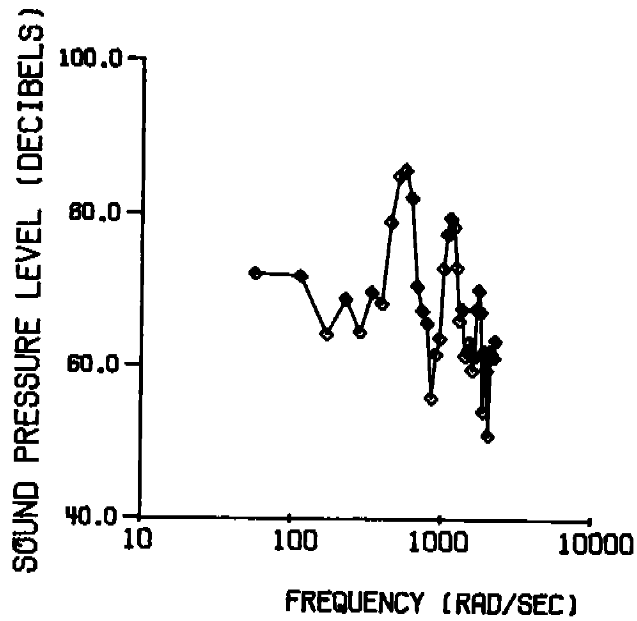

Figure 3. Sound Spectrum for Open Valve

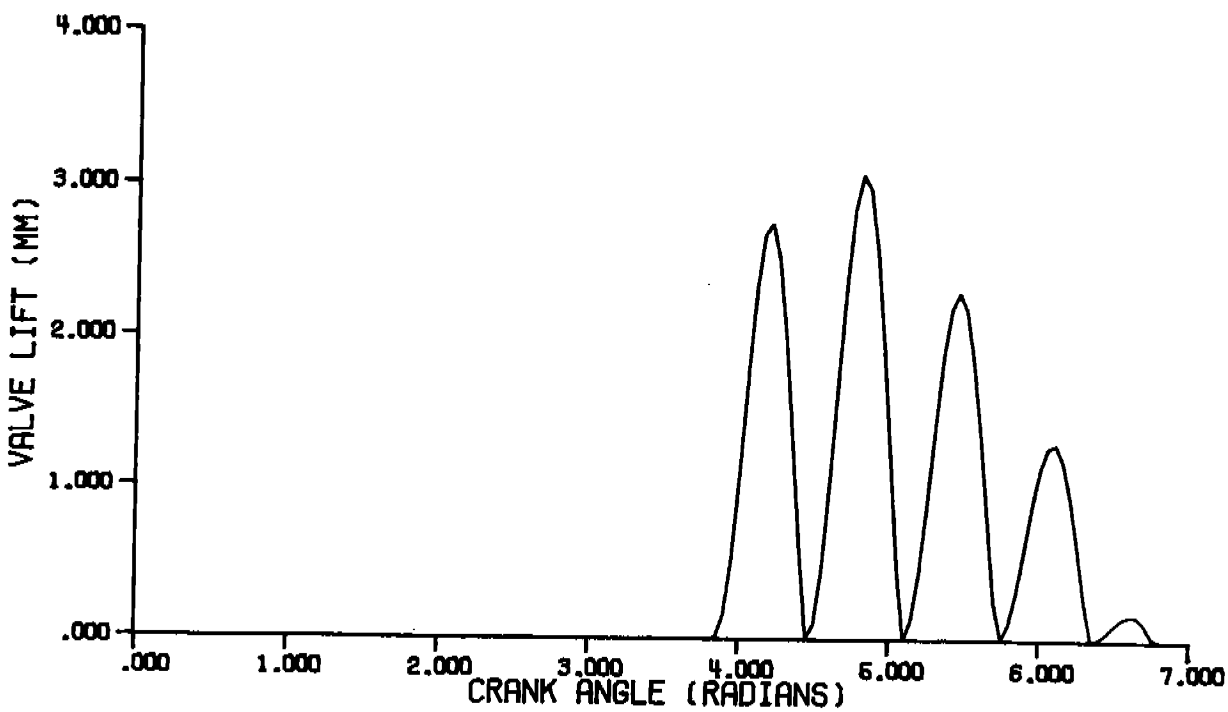

Figure 4. Suction Valve Displacement Without Stop

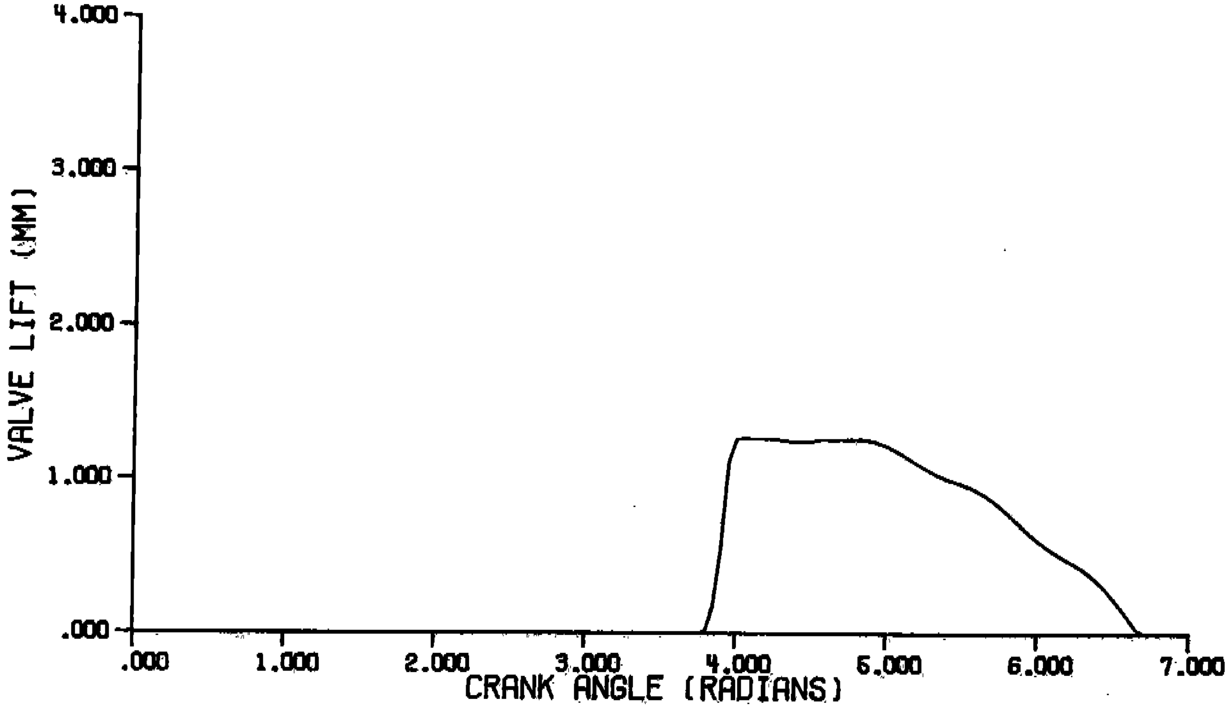

Figure 5. Suction Valve Displacement with Valve Stop

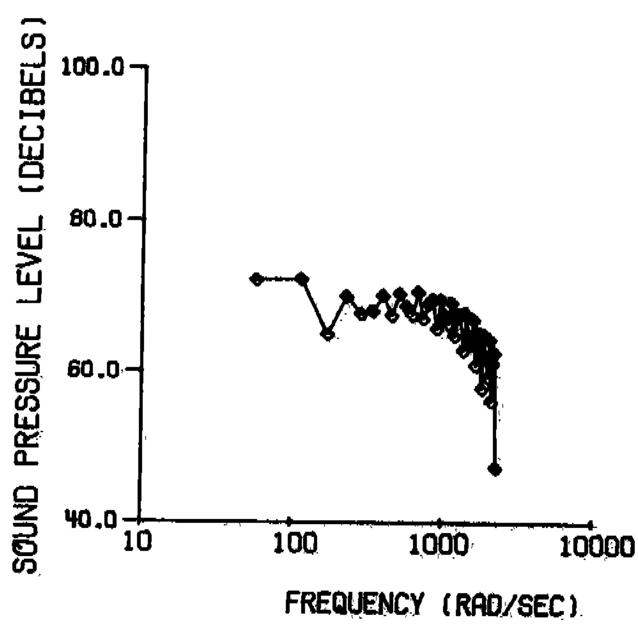

Figure 6. Sound Spectrum for Open Valve with Stop

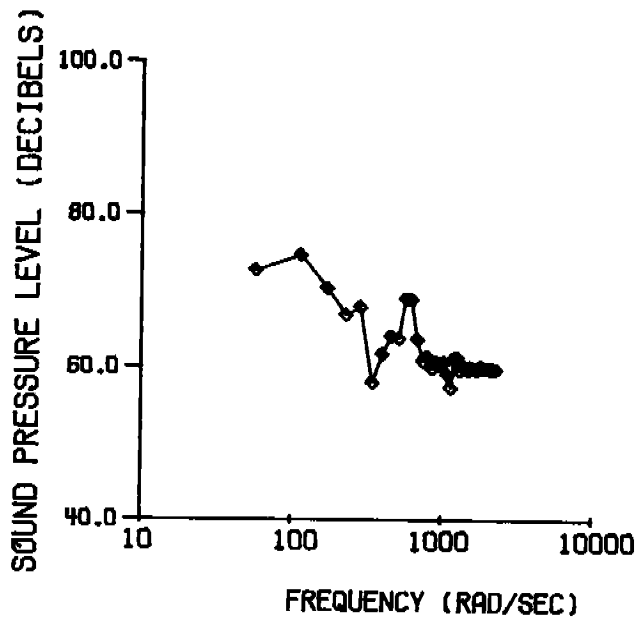

Figure 7. Sound Spectrum for Valve without Stop but with Suction Muffler

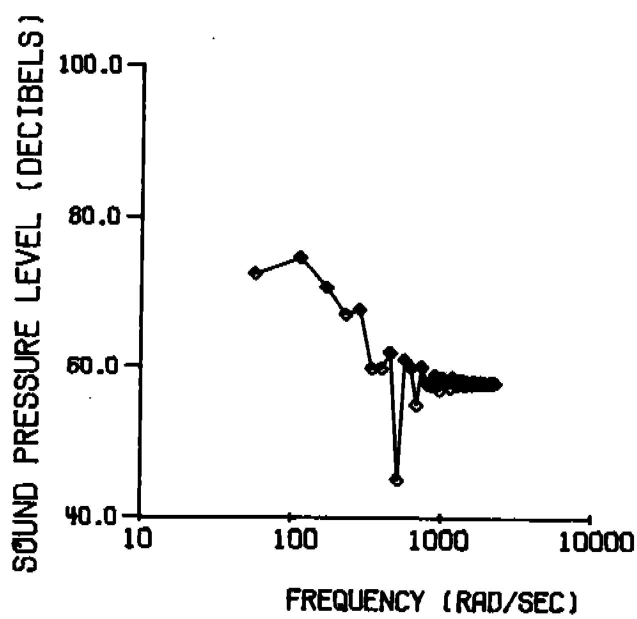

Figure 8. Sound Spectrum for Valve with Both Stop and Suction Muffler